Art Unit 3502

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Appeal No. 93-3313

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BOARD OF PATENT APPEALS
AND INTERFERENCES

HEARD:
October 7, 1993

UNITED STATES PATENT AND TRADEMARK OFFICE

BEFORE THE BOARD OF PATENT APPEALS
AND INTERFERENCES

Ex parte Satoshi Kono,
Shizuaki Hidaka and
Tetsu Takahashi

- - -

Application for Patent filed February 27, 1990, Serial No. 485,659. Flywheel For Power Transmission System For Transmitting Engine Torque To Driven Unit.

Ronald P. Kananen for appellants.

Primary Examiner - Vinh T. Luong.

Before McCandlish, Parsons and Lyddane, Administrative Patent Judges.

McCandlish, Administrative Patent Judge.

This appeal is from the examiner's rejection of claims 11 through 16 and 18 under 35 USC 103. All of the remaining

claims have either been cancelled or have been withdrawn as being directed to a nonelected invention.

The subject matter here claimed is a flywheel for transmitting engine torque to a driven unit. Appellants summarize the invention as follows:

> The invention features a flywheel which comprises a flywheel body which is connected with an engine crankshaft by way of an elastic or resilient plate. The elastic member has a torsional rigidity sufficient to effectively transmit driving power or torque. The elastic member further features a rigidity in the axial direction (viz., axial with respect to the engine crankshaft) which is small enough to shift the torsional vibration resonance [sic, resonant] frequency out of a predetermined frequency band which tends to occur during vehicular acceleration, while simultaneously ensuring the proper [clutch] disengagement characteristics. (brief, page 2)

The predetermined frequency band mentioned in the foregoing quotation is disclosed in appellants' application to be in a range extending from 200 Hz to 500 Hz. If the resonant frequency of the "bending vibration in an axial direction of the crankshaft" (specification, page 1) is within this frequency band, objectionable noise is produced in the engine compartment and thus in the vehicle passenger compartment. Shifting the

resonant frequency of the bending vibration level to a value lower than 200 Hz has the effect of alleviating the noise problem. The resonant frequency is determined by the axial rigidity of the elastic plate 2. In this regard, reducing the axial rigidity, that is, making the elastic plate more resilient or flexible in the axial direction, has an effect of reducing the value of the resonant frequency of the flexural or bending vibration in the axial direction.

Independent claims 11 and 16 recite that the axial rigidity of the elastic plate is in the range of 600 kg/mm to 2200 kg/mm.[1] The upper limit of 2200 kg/mm is depicted by the curve A1 in Figure 2 of the application drawings. The resonant frequency for curve A1 is shown to lie very close to the lower 200 Hz limit of the objectionable frequency band mentioned <u>supra</u>.

According to appellants' specification, the lower limit of 600 kg/mm is significant in that if the axial rigidity of the elastic plate is reduced to a value lower than this limit, problems with clutch engagement and disengagement will occur.

Another claimed feature of the invention relates to what is described as an "axial run-out" of the clutch engaging

[1] Although not expressly stated in appellants' specification, it appears that these values of force are intended to mean that for the upper limit of the range, the elastic plate requires a force of 2200 kg to flex through a distance of 1 millimeter.

surface of the flywheel body. This expression is understood to mean wobble of the clutch-engaging flywheel face in the axial direction. As stated in appellants' specification, the reduction of axial run-out to a value not exceeding 0.1 mm reduces fore and aft vibrations to an acceptable level.

Claims 11, 12 and 16 recite the foregoing axial rigidity range of 600 kg/mm to 2200 kg/mm but contain no limitation pertaining to the axial run-out. Claims 14 and 18, on the other hand, contain a limitation regarding the axial run-out of the clutch engaging flywheel surface, but are not limited to any particular value of axial rigidity for the elastic plate.[2] Claims 13 and 15 contain limitations pertaining to both the axial rigidity of the elastic plate and the axial run-out of the clutch-engaging flywheel.

A copy of independent claims 11, 14, 16 and 18, as these claims appear in the appendix to appellants' brief, is appended to this decision.

In rejecting the appealed claims, the examiner relies upon the following references:

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[2] Appellants' summary of the invention, as set forth in the brief, is incomplete in that there is no mention of the axial run-out feature.

Numata et al. (Numata) (Japanese Patent Publication) (Translation attached)	57-058542	Dec. 10, 1982
Cucinotta et al.[3] (Cucinotta) (European Patent Application)	0 048 563	Mar. 31, 1982

All of the appealed claims stand rejected under 35 USC 103 as being unpatentable over the Numata reference. With particular regard to the appealed claims relating to the axial run-out feature, the examiner is understood to also rely upon appellants' prior art admissions set forth in lines 2 through 36 on page 2 of appellants' specification. See page 3 of the answer. Claims 11 through 15 are additionally rejected under 35 USC 103 as being unpatentable over the Cucinotta reference.

Reference is made to the examiner's answer for details of these rejections. The final rejection of the appealed claims under 35 USC 112 ¶ 2 has been withdrawn in view of appellants' amendment after the final rejection.

In support of patentability of the appealed claims over Numata, the only limitations argued as a difference over this reference in the argument section of appellants' brief is that

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[3] The examiner incorrectly referred to this reference by the inventor's first name.

the Japanese reference "neither discloses the claimed range of axial rigidity, nor the claimed [axial] run-out limit" (main brief, page 6).[4]

We have carefully considered the issues raised by the examiner's rejection based on Numata together with the examiner's remarks and appellants' arguments. As a result, we conclude that the rejection based on this reference together with appellants' prior art admissions set forth on pages 1 and 2 of the specification is sustainable, although not for all of the reasons stated by the examiner.

As is evident from the accompanying translation, Numata is clearly concerned with the noise problem that emanates from what is described as the bending vibration level of the crankshaft as set forth, for example, on pages 2, 4, 5 and 6 of the accompanying translation. The bending vibrations are

[4] According to 37 CFR 1.192(c)(6), appellants' contentions supporting patentability and particularly the specification of errors of the examiner's § 103 rejection shall be specified in the argument section of the brief. The intent of this rule is to preclude placement of arguments in other sections of the brief where they may be overlooked. We mention this only because certain arguments supporting patentability appear to have improperly been incorporated into the section concerning the grouping of claims. We also note with interest that the statement concerning the grouping of claims is inadequate if for no other reason other than that there is no statement of grouping as to each ground of rejection.

referred to as being in the shaft direction as noted, for example, on pages 4 and 5 of the accompanying translation. As set forth on page 2 of the accompanying translation, Numata recognizes that a problem with unpleasant noise occurs when the bending vibrations are about or above 210 Hz or cycles per second as the value is described in the Japanese specification.

To overcome this noise problem, Numata teaches the art to reduce the resonant frequency of the bending vibration level below the value of 210 Hz by providing an elastic plate between the crankshaft and the flywheel and by providing the elastic plate with a low bending rigidity in the direction of the shaft as set forth, inter alia, on pages 2 through 4 of the Japanese reference. The characterization of the elastic plate as having a "bending rigidity" or more specifically a "low bending rigidity" is expressly stated to be in the direction of the crankshaft as set forth on pages 2, 3 and 4 of the accompanying translation of the Japanese reference.

Thus, Numata's reference to the bending rigidity of the elastic plate obviously corresponds to appellants' reference to the axial rigidity of the elastic plate as set forth in appellant's claims and specification. This interpretation of Numata is supported by appellants' own discussion of Numata on pages 1 and 2 of the specification.

According to appellants' description of Numata, the elastic plate in this reference has "a rigidity in the axial direction small enough for shifting a resonance [sic] frequency of the bending vibration out of a frequency band of a forced vibration which results during the most frequently used engine speed (4000 rpm) so as to overcome the above-noted [noise] problem" (specification, page 2).

Appellants do not chose to make any distinction between the lower frequency band limit of 210 Hz in Numata and the lower limit of 200 Hz in appellants' specification. Thus, an elastic plate having an axial rigidity corresponding to a resonant frequency of less than 210 Hz would approach if not fall within appellants' claimed range of axial rigidities. One of ordinary skill in the art would interpret the teachings of Numata as requiring an axial rigidity for the elastic plate which is significantly lower than the value associated with the lower limit of 210 Hz, thus implicitly providing a teaching of selecting an axial rigidity within the claimed range.

The fact that Numata may not disclose the particular <u>range</u> claimed by appellants for the axial rigidity of the elastic plate is not controlling. Instead, it has long been held that the disclosure in the prior art of <u>any value</u> within a claimed range is an anticipation of that claimed range. See, for

example, In re Wertheim, 541 F.2d 257, 191 USPQ 90 (CCPA 1976) and Titanium Metals Corp. of America v. Banner, 778 F.2d 687, 227 USPQ 773 (Fed. Cir. 1985).

Thus, given our analysis of the Japanese reference, Numata contains at least an implicit teaching of an axial rigidity falling within the range as defined in claims 11, 15 and 16. Even if it is assumed arguendo that the resonant frequency associated with the axial rigidity for the elastic plate in Numata is not reduced low enough to fall within the claimed range of axial rigidities, it would appear to us that the variation is so slight that it would have been obvious to one of ordinary skill in the art. See In re Woodruff, 919 F.2d 1575, 16 USPQ2d 1934 (Fed. Cir. 1990). See also Titanium Metals Corp. of America v. Banner, supra.

At oral argument, appellants' counsel was understood to contend that the axial rigidity of the elastic plate in the Numata reference was actually less than the claimed lower limit of 600 kg/mm. Aside from the fact that this argument was not made in either of appellants' briefs and finds no support in the record before us, it is well settled that arguments of counsel do not take the place of evidence. See, for example, In re DeBlauwe, 736 F.2d 699, 222 USPQ 191 (Fed. Cir. 1984).

In view of the foregoing, the subject matter of claim 11 would have been obvious from the teachings of Numata if not anticipated by Numata. We will therefore sustain the examiner's rejection of claim 11 based on the Numata reference. We will also sustain the examiner's rejection of dependent claim 12 based on Numata inasmuch as this claim has not been argued separately of claim 11. See In re Nielson, 816 F.2d 1567, 2 USPQ2d 1525 (Fed. Cir. 1987) and In re Burckel, 592 F.2d 1175, 201 USPQ 67 (CCPA 1979).

Turning now to the § 103 rejection of claim 16 based on Numata, appellants argue that in addition to lacking a disclosure of the claimed range for the axial rigidity of the elastic plate, the Japanese reference does not have a reinforcing member defining a space between the elastic plate and the flywheel body. We disagree.

The space defined in claim 16 is recited to lie between the elastic plate and the flywheel body. This space, therefore, is the one lying between the flywheel surface 5f and the confronting face of the elastic plate 2 in appellants' embodiment of Figure 1. This space, however, is not defined in its entirety by the reinforcing member 4 as appellants' arguments seem to suggest. Instead, the reinforcing member 4 merely delimits the claimed space to define the space in cooperation with the confronting faces of the elastic plate 2 and the flywheel body 5.

In the embodiment shown in Figure 4 of Numata, the plate member 24 also delimits the space between the elastic plate 3 and the flywheel body 9 in that it covers and thus provides a boundary for the opening 2 through the elastic plate 3. As such, Numata's plate member 24 also defines the space between the elastic plate 3 and the flywheel body 9 in cooperation with the elastic plate and the flywheel in the same sense that appellants' reinforcing member defines the space in question. This space is identified by the examiner in Exhibit I which is attached to the examiner's answer.

Furthermore, contrary to appellants' argument, Numata's plate member 24 constitutes a reinforcing member in that it performs a reinforcing function by providing a backing for the elastic plate 3. We are therefore satisfied that claim 16 would have been obvious from if not anticipated by the Numata reference. Accordingly, we will also sustain the examiner's rejection of claim 16 based on Numata.

Turning now to the rejection of claim 14 based on the Numata reference, the examiner contends on page 3 of the answer that according to appellants' prior art admissions on page 2 of the specification, the clutch-engaging face of the flywheel is known to have an axial run-out. In support of patentability,

appellants argue as on page 6 of the main brief that the Japanese reference does not teach the claimed axial run-out limit. Appellants additionally argue as follows:

> The applicant [sic, appellants] also notes [sic, note] that the Examiner considers there to be some form of "admission" concerning the prior art on pages 1 and 2 of the specification, and has possibly drawn upon the applicant's [sic, appellants'] own disclosure for directions as how the above mentioned axial rigidity and axial run-off parameters should be set.
>
> Firstly, the applicants' [sic, appellants'] own disclosure is basically not available as a reference against the claims. (answer, page 7)

Admittedly, an inventor's own invention may not be prior art against him absent a statutory bar. In re Katz, 687 F.2d 450, 215 USPQ 14 (CCPA 1982). However, as appellants well know, the examiner is not relying upon appellants' own work, but instead is relying upon prior art admissions. In this regard, it is well settled that in ex parte examination of a patent application, admissions of prior art by an applicant are considered for any purpose including evidence of obviousness under § 103. See, inter alia, In re Nomiya, 509 F.2d 566, 184 USPQ 607 (CCPA 1975), In re Hellsund, 474 F.2d 1307, 177 USPQ 170 (CCPA 1973) and In re Garfinkel, 437 F.2d 1000, 168 USPQ 659 (CCPA 1971).

Appellants' discussion on pages 1 and 2 of the specification under the subheading "Description of the Background Art" is considered to be a discussion of prior art known to appellants. This introductory portion of appellants' specification discusses the axial run-out in terms of what appellants refer to as "Background Art" on page 2 of the specification as follows:

> Further, in the background art, when the flywheel is rotated, an axial run-out occurs on an engaging surface of the flywheel with a clutch facing of a clutch disc provided adjacent to the flywheel, due to a processing error and an assembling error of the elastic plate and the flywheel. Accordingly, when the clutch is engaged, a vibration is generated by a combination of the run-out of the engaging surface of the flywheel and the torque fluctuation of the engine, which is amplified by a vibration generated by the combustion in the engine cylinders and corresponding movements of associated members so as to cause a fore and aft vibration of the vehicle floor. Such vibration is uncomfortable for the driver and passengers in the vehicle compartment.

As we understand appellants' discussion of background art as quoted supra, axial run-out of the clutch-engaging face of the flywheel was known to cause the objectionable fore and aft vibration of the vehicle floor. It is also apparent that the axial run-out could be reduced by known manufacturing processes

to eliminate errors. Given this state of the prior art, it would have been obvious to reduce the axial run-out of the clutch-engaging surface of the flywheel if the level of the fore and aft vibration of the vehicle floor were considered to be objectionable. Certainly, skill in the art is presumed, not the converse. In re Sovish, 769 F.2d 738, 226 USPQ 771 (Fed. Cir. 1985).

The extent to which the fore and aft vibrations are reduced by reducing the axial run-out of the clutch-engaging surface of the flywheel would have been a mere matter of design choice, depending upon the extent of comfort to be accorded to passages in the vehicle and upon possible increased manufacturing cost in reducing the axial run-out. In this regard, it has been held that where the advantages and disadvantages of an expedient are known, the expedient would have been an obvious matter of design choice. See In re Heinrich, 268 F.2d 753, 122 USPQ 388 (CCPA 1959).

For these reasons, we will sustain the examiner's § 103 rejection of claim 14 based on the Numata reference. Likewise, we will also sustain the § 103 rejection of dependent claims 13 and 15 based on the Numata reference for the reasons discussed supra in our review of the rejection of claims 11 and 14.

Finally, we will also sustain the examiner's § 103 rejection of claim 18 based on the Numata reference for the reasons set forth in our affirmance of the rejection of claims 14 and 16.

Inasmuch as our reasons affirming the examiner's rejection of the appealed claims based on the Numata reference differ from those advanced by the examiner and inasmuch as we may have relied more heavily on appellants' prior art admissions particularly as set forth on page 2 of the specification, we herewith designate our affirmance of claims 11 through 16 and 18 based on the Numata reference as a new ground of rejection under 37 CFR 1.196(b). It should be understood that in this new ground of rejection, we relied not just on the Numata reference alone, but on the combination of the Numata reference with the prior art admissions set forth on page 2 of appellants' specification, particularly the prior art admissions concerning axial run-out commencing in the last paragraph on page 2 of the specification.

We cannot, however, sustain the examiner's § 103 rejection of claims 11 through 15 based on the Cucinotta reference alone. In the first place, element 11 in Cucinotta's power transmission apparatus is described as a cover, not an elastic plate. In the second place, Cucinotta does not recognize any significance in the axial rigidity of cover 11. Finally,

when Cucinotta is considered alone, there is no teaching concerning axial run-out or the effect thereof on vibrations in the vehicle floor.

In summary, the examiner's decision rejecting appealed claims 11 through 16 and 18 on the Japanese Numata reference is affirmed, while his decision rejecting appealed claims 11 through 15 on the Cucinotta reference is reversed. Additionally, our affirmance of the rejection based on the Japanese reference has been designated as a new ground of rejection under the 37 CFR 1.196(b).

Any request for reconsideration or modification of this decision by the Board of Patent Appeals and Interferences based upon the same record must be filed within one month from the date hereof (37 CFR 1.197).

With respect to the new rejection under 37 CFR 1.196(b), should appellants elect the alternate option under that rule to prosecution further before the Primary Examiner by way of amendment or showing of facts, or both, not previously of record, a shortened statutory period for making such response is hereby set to expire two months from the date of this decision. In the event appellants elect this alternate option, in order to preserve the right to seek review under 35 USC 141 or 145 with respect to the affirmed rejection, the effective date of the

affirmance is deferred until conclusion of the prosecution before the examiner unless, as a mere incident to the limited prosecution, the affirmed rejection is overcome.

If the appellants elect prosecution before the examiner and this does not result in allowance of the application, abandonment or a second appeal, this case should be returned to us for final action on the affirmed rejection, including any timely request for reconsideration thereof.

No time period for taking any subsequent action in connection with this appeal may be extended under 37 CFR 1.136(a). See the final rule notice, 54 F.R. 29548 (July 13, 1989), 1105 O.G. 5 (August 1, 1989).

<u>AFFIRMED</u>
<u>37 CFR 1.196(b)</u>

Harrison E. McCandlish Administrative Patent Judge	BOARD OF PATENT APPEALS AND INTERFERENCES
Marion Parsons, Jr. Administrative Patent Judge	
William E. Lyddane Administrative Patent Judge	

Appeal No. 93-3313

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Appeal No. 93-3313

APPENDIX

11. A flywheel for a power transmission system for transmitting engine torque to a driven unit, comprising:

an elastic plate secured to a crankshaft to rotate therewith; and

a flywheel body secured to said elastic plate and having an engageable surface which is engageable with a clutch disc,

said elastic plate having an axial rigidity in the range of 600 kg/mm to 2200 kg/mm so as to ensure transmission of engine torque to said driven unit, while decreasing noise produced by a bending vibration of said crankshaft.

14. A flywheel for a power transmission system for transmitting engine torque to a driven unit, comprising:

an elastic plate secured to a crankshaft to rotate therewith; and

a flywheel body secured to said elastic plate and having an engageable surface which is engageable with a clutch disc,

said engageable surface having an axial run-out which is equal to or less than 0.1 mm for ensuring a smooth engagement with said clutch disc.

16. A flywheel for a power transmission system for transmitting engine torque to a driven unit, comprising:

an elastic plate secured to a crankshaft to rotate therewith; and

a flywheel body secured to said elastic plate and having an engageable surface which is engageable with a clutch disc;

a reinforcing member for reinforcing said elastic plate at a portion of said elastic plate which is secured to said crankshaft, said reinforcing member defining a space between said elastic plate and said flywheel body,

said elastic plate having an axial rigidity in the range of 600 kg/mm to 2200 kg/mm so as to ensure transmission of engine torque to said driven unit, while decreasing noise produced by a bending vibration of said crankshaft.

18. A flywheel for a power transmission system for transmitting engine torque to a driven unit, comprising:

an elastic plate secured to a crankshaft to rotate therewith;

a flywheel body secured to said elastic plate and having an engageable surface which is engageable with a clutch disc; and

a reinforcing member for reinforcing said elastic plate at a portion of said elastic plate which is secured to said crankshaft, said reinforcing member defining a space between said elastic plate and said flywheel body,

said engageable surface having an axial run-out which is equal to or less than 0.1 mm for ensuring a smooth engagement with said clutch disc.

Japanese Patent Publication No. 57-058542, published December 10, 1982; Kokai Publication No. 53-32524, published March 27, 1978; application No. 51-107492, filed September 7, 1976; inventors Hoji NUMATA, et. al.; assignee, Mitsubishi Jidosha Kogyo KK

AUTOMOBILE TRANSMISSION MECHANISM

Claim:

An automobile transmission mechanism that couples the crankshaft of an internal combustion engine and a flywheel [supported by the main drive shaft through a bearing and coupled with the main drive shaft through a clutch] through a highly rigid elastic sheet so as to provide a tight coupling in the direction of the rotational twist, while on the other hand having low bending rigidity in the shaft direction.

Detailed Description of the Invention:

This invention concerns an improvement to an automobile transmission mechanism which transmits power through the flywheel and clutch by the crankshaft of an internal combustion engine.

Commonly, three types of vibration --- twist vibration, bending vibration and vertical vibration --- are considered in making the crankshaft of an automobile's internal combustion engine. A rigid body flywheel is especially required in order to reduce the twist vibration and to obtain smooth rotation. However, the dynamic properties of the crankshaft system, including this flywheel, can create problems. More specifically, when determining the shape of the flywheel so as to provide the best properties to deal with twist vibration [which originates by the mass, inertial

moment and twist rigidity], speed changes and starting, etc., many automobiles easily develop a murmuring sound inside of the engine. This is caused by the bending vibration of the crankshaft when the engine rpm reaches about 4,000, or, in terms of speed, when the automobile reaches about 100 km/h. A worst case situation may result in broken parts caused by screws loosened by the vibration.

The characteristic number of crankshaft bending vibrations is about 210 c/s. Therefore, if this characteristic number of bending vibrations of the crankshaft system can be removed from the above mentioned range of rpm and speed, yet without changing the dynamic properties of the crankshaft rotation, the unpleasant noise caused by the bending and twisting vibrations of the crankshaft system can be reduced.

The main objective of this invention is to offer an automobile transmission that releases the bending resonance point of the engine's crankshaft from the speed and rpm range where the engine noise is generated by the bending vibration. This reduces the noise and possible part breakage.

This invention's above mentioned objective is achieved by creating an automobile transmission mechanism that couples the crankshaft of an internal combustion engine and a flywheel [supported by the main drive shaft through a bearing and coupled with the main drive shaft through a clutch] through a highly rigid elastic sheet so as to provide a tight coupling in the direction of the rotational twist, while on the other hand having low bending rigidity in the direction of the shaft.

Details of this invention are explained below with reference to the accompanying practical examples and drawings.

In each drawing, essentially the same symbols are used for the same parts.

In the first example of this invention, as shown in Figure 1, an elastic disc (3) is affixed to the end of an automobile engine's crankshaft (1). This disc (3) is highly rigid so as to result in an essentially tight coupling in the direction of the rotational twist. The low value of the bending rigidity in the direction of the shaft is set by adjusting its thickness.

The end of the main drive shaft (6) [supported by a pivot inside of the clutch housing (4) through a bearing (5)] is supported by pivot at a pilot bearing (7) [installed at the centripetal section of the crankshaft (1)].

A flywheel (9) is supported by a pivot on the main drive shaft (6) through a bearing (8). Simultaneously, a spline hub (12) of a commonly shaped clutch disc (11) is inlaid so as to slide in direction of the shaft line over the spline shaft section (10) of the shaft (6).

A clutch cover (14) is affixed at the external periphery of the flywheel (9) by a bolt (13). A common diaphragm spring (15) is installed on the clutch cover (14). The external rim of the spring (15) presses the protruding rim of a pressure plate (16). This pressure plate (16) then presses the clutch disc (11) and the clutch coupling is achieved.

Item (17) is a front cover; (18) is a front cover sleeve; and

(19) is a release bearing.

Each external peripheral rim sections of the flywheel (9) and the elastic disc (3) is affixed by several bolts (20).

With the above mentioned structure, there is high rigidity so as to create essentially a right coupling to the direction of the rotational twist of the elastic disc (3). As a result, not only is there a decrease in the dynamic properties which are generated by the rotational twisting of the crankshaft (1), but the bending rigidity of the crankshaft can be greatly reduced by providing low bending rigidity to the elastic disc (3). The bending resonance point of the system can be greatly reduced to a point lower than 210 c/s at 4,000 rpm range where the problem is generated. As a result, the engine noise and parts breakage can be effectively reduced.

In the above mentioned example, by installing the flywheel (9) over the main drive shaft (6) through a common single ball bearing, fine displacement in the direction of the flywheel (9) bending is allowed. Thus, bending in the direction of the shaft of the elastic disc (3) becomes easy and the characteristic number of bending vibrations of the crankshaft can be sufficiently reduced.

The above first example uses an existing common structure in which the structure of the main drive shaft (6) is supported by the crankshaft (7). However, this invention is not restricted to only this above example. For example, as shown in the modified example in Figure 2, instead of using a structure supported by the pilot bearing (7) of the main drive shaft (6), it can be a structure in

which the support bearing (20) of the main drive shaft (6) is provided at an area close to the flywheel (9) of the front cover (17') which is affixed to the clutch housing (4) and externally located on the main drive shaft (6). In this case, the bending in the direction of the shaft of the elastic disc (3) becomes easy as a result of the low vibration of the main drive shaft (6). A large slanting movement of the flywheel caused by gyro-moment is prevented by the support of double ball bearings (8'). Another modified example as shown in Figure 3 has a structure in which a spherical bearing (21) is formed at the end surface of the crankshaft (1) in order to prevent whirling of the main drive shaft (6). A spherical U-shaped section (22) is formed at the end surface of the crankshaft (1). A globular section (23) is formed at the flywheel (9).

A second example of this invention, as shown in Figures 4 and 5, is explained below. In this example, the guide stopper plate (24) and the elastic disc (3) are affixed at the end surface of the crankshaft (1) by a bolt (2) under laminated conditions. A ball member (25) which forms a spherical bearing (21') is affixed to the end surface center section of the crankshaft (1). A spherical U-shaped section (26) which forms a spherical bearing (21') is formed at the flywheel (9) by making contact with the ball member (25). The guide stopper plate (24) is formed with a plate-like curved shape so as to gradually increase the interval to the elastic disc (3) along the outward radius direction.

In this example, multiple round holes (27) are also provided

on the elastic disc (3) with space between them in the peripheral direction in order to make the bending rigidity in the direction of the shaft of the elastic disc (3) closer to the established value. The bending rigidity can be easily adjusted by providing holes of a suitable size and number on the elastic disc (3) at a proper location.

In the above second example, when the flywheel (9) attempts a large slanting movement caused by a gyro-moment, it is caught by the guide stopper plate (24) and the extreme slanting can be prevented.

In this case, the interval between the guide stopper plate (24) and the elastic disc (3) is made so as to gradually increase in the radius direction. Therefore, both gradually make contact with each other by facing in the radius direction. Thus, there is no engine noise or parts damage.

A great decrease in the bending rigidity of the elastic disc (3) is made possible by providing the above mentioned guide stopper plate (24). Consequently, the characteristic number of bending vibrations of the crankshaft can be sufficiently reduced.

Simple Explanation of Drawings:

Figure 1 is a cross-sectional view of the first example of this invention. Figure 2 is a cross-sectional view showing a modification of the example in Figure 1. Figure 3 is a cross-sectional view showing another modification of the first example. Figure 4 is a cross-sectional view of a second example of this invention. Figure 5 is a side view of the second example.

1... crankshaft

2... bolt

3... elastic disc

4... clutch housing

5,7,8,8'... bearing

6... main drive shaft

9... flywheel

11...clutch disc

17, 17'... front cover

20...bolt

21,21'...spherical bearing

24...guide stopper plate

27... round hole

U. S. Patent and Trademark Office
January 11, 1993
Y.O.